Laboratory Air Handling Unit System

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ABSTRACT

An innovative AHU system is presented in this paper. The proposed AHU system is called a Laboratory Air Handling Unit (LAHU[1]) system since it is most suitable for the buildings where one section (laboratory) has 100% exhaust while the other section (office) allows air to circulate. The LAHU system uses less thermal energy, provides better indoor air quality, and requires less system capacity than traditional AHU systems.

INTRODUCTION

Modern research buildings often contain both laboratories and office space. Since the laboratory section cannot use recycled air, designs traditionally use 100% outside air.

In the past 20 years, much research work has concentrated on decreasing energy consumption in laboratory buildings. Several energy conservation measures and designs have been developed. Exhaust air is used to pre-condition outside air through a heat exchanger during both winter and summer months [Barker 1994; Bard 1994]. The heat recovery greatly decreases mechanical cooling and heating energy consumption. Outside air is significantly decreased when variable air volume fume hoods are used [Neuman and Rousseau 1986; Davis and Benjamin 1987]. A VAV system may consume 50% less energy than a constant volume system. Today, it is a common practice to use separated systems in laboratory and office sections in research buildings. A dedicated AHU for an office section allows shutdown on weekends and at nights. Consequently, the overall energy consumption is decreased.

Taking an integrated energy analysis approach, a Laboratory Air Handling Unit (LAHU) system is developed in this paper. The LAHU provides better indoor air quality for the office section but uses significantly less thermal energy. The potential energy models for the LAHU system are developed. The optimal energy savings and outside air intake ratio are evaluated and analyzed.

SYSTEM MODELS

This section presents energy consumption models for both the base system and the LAHU system. The potential energy savings is determined by taking the difference of the base system and the LAHU system under the same load conditions.

Base System Model

Figure 1 shows a schematic diagram of a conventional air handling system for laboratory buildings (base system). AHU1 serves the office section and AHU2 serves the laboratory section. AHU1 uses recycled air along with the minimum outside air required to make up the total supply air requirement. AHU2 uses 100% outside air.

Taking an integrated approach, the energy balance, air mass balance and moisture balance are given below for the entire building.

$$\dot{Q}_i + \dot{Q}_h + \dot{Q}_{hg} = \dot{Q}_r + \dot{Q}_C + \dot{Q}_e + \dot{Q}_{eh} + \dot{Q}_{env} \qquad (1)$$

Where

$$\dot{Q}_i = \sum_{j=1}^{2} \dot{Q}_{i,j}$$

$$\dot{Q}_h = \sum_{j=1}^{2} \dot{Q}_{h,j}$$

$$\dot{Q}_{hg} = \sum_{j=1}^{2} \dot{Q}_{hg,j}$$

$$\dot{Q}_C = \sum_{j=1}^{2} \dot{Q}_{C,j}$$

$$\dot{Q}_e = \sum_{j=1}^{2} \dot{Q}_{e,j}$$

[1] Patent Pending

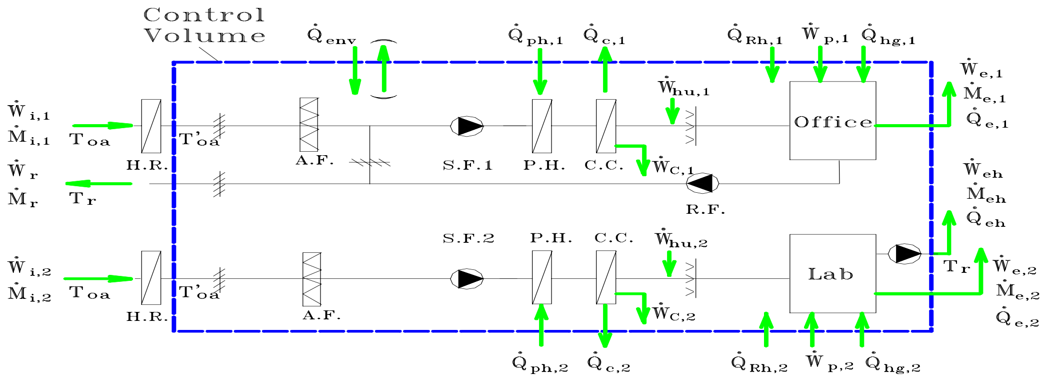

Figure 1. Schematic diagram of air handling unit system for base system

$$\dot{M}_i = \dot{M}_r + \dot{M}_e + \dot{M}_{eh} \qquad (2)$$

Where

$$\dot{M}_i = \sum_{j=1}^{2} \dot{M}_{i,j}$$

$$\dot{M}_e = \sum_{j=1}^{2} \dot{M}_{e,j}$$

$$\dot{W}_i + \dot{W}_p + \dot{W}_{hu} = \dot{W}_C + \dot{W}_e + \dot{W}_{eh} \qquad (3)$$

Where

$$\dot{W}_i = \sum_{j=1}^{2} \dot{W}_{i,j}$$

$$\dot{W}_p = \sum_{j=1}^{2} \dot{W}_{p,j}$$

$$\dot{W}_{hu} = \sum_{j=1}^{2} \dot{W}_{hu,j}$$

$$\dot{W}_C = \sum_{j=1}^{2} \dot{W}_{C,j}$$

$$\dot{W}_e = \sum_{j=1}^{2} \dot{W}_{e,j}$$

Thermal energy consumption is the sum of heating and cooling energy consumption. The thermal energy consumption of equation (1) is rewritten in the form of equation (4), in order to facilitate the potential energy savings calculation later.

$$\dot{Q}_C + \dot{Q}_h = 2\,\dot{Q}_C + \dot{Q}_r + \dot{Q}_e + \dot{Q}_{eh} + \dot{Q}_{env} - \dot{Q}_i - \dot{Q}_{hg} \qquad (4)$$

The LAHU system has no impact on the fan power consumption. This study assumes zero infiltration and zero humidification energy consumption.

LAHU system

Figure 2 shows a schematic diagram of the LAHU system. The LAHU has two supply fans and one return fan. Fan 1(S.F.1) supplies air to the office section. Fan 2 (S.F.2) supplies air to the laboratory section. The return fan recirculates the return air from the office section to either supply fan 1 or 2, or both. Return air can be sent either upstream or downstream of cooling coil 2, or in both directions.

The energy balance, air mass balance and moisture balance equations are listed below for the entire building:

$$\dot{Q}_C^{'} + \dot{Q}_h^{'} = 2\,\dot{Q}_C^{'} + \dot{Q}_r^{'} + \dot{Q}_e + \dot{Q}_{eh} + \dot{Q}_{env} - \dot{Q}_i^{'} - \dot{Q}_{hg} \qquad (5)$$

Where

$$\dot{Q}_C^{'} = \sum_{j=1}^{2} \dot{Q}_{C,j}^{'}$$

$$\dot{Q}_h^{'} = \sum_{j=1}^{2} \dot{Q}_{h,j}^{'}$$

$$\dot{Q}_i^{'} = \sum_{j=1}^{2} \dot{Q}_{i,j}^{'}$$

$$\dot{M}_i^{'} = \dot{M}_r^{'} + \dot{M}_e + \dot{M}_{eh} \qquad (6)$$

Where

$$\dot{M}_i^{'} = \sum_{j=1}^{2} \dot{M}_{i,j}^{'}$$

$$\dot{W}_i^{'} + \dot{W}_p + \dot{W}_{hu}^{'} = \dot{W}_C^{'} + \dot{W}_e + \dot{W}_{eh} \qquad (7)$$

Where

$$\dot{W}_i^{'} = \sum_{j=1}^{2} \dot{W}_{i,j}^{'}$$

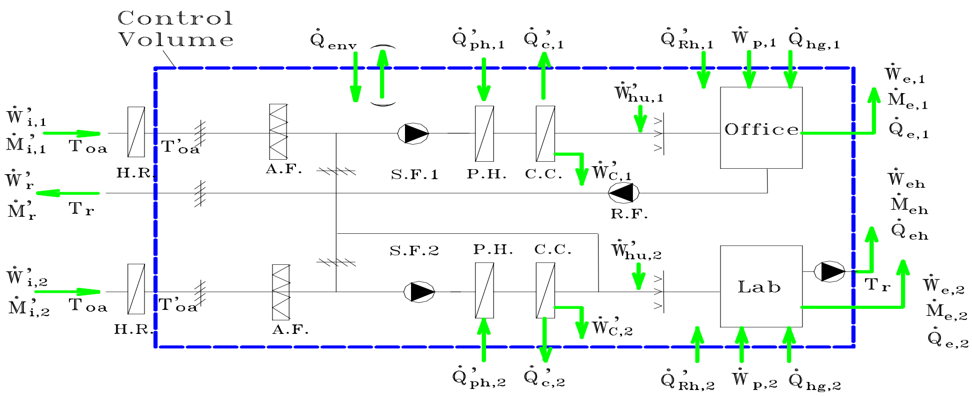

Figure 2. Schematic diagram of air handling unit system for LAHU system

$$\dot{W}_{hu}^{'} = \sum_{j=1}^{2} \dot{W}_{hu,j}^{'}$$

$$\dot{W}_{C}^{'} = \sum_{j=1}^{2} \dot{W}_{C,j}^{'}$$

The energy savings is defined as the energy consumption difference between the base system and the LAHU system. If the laboratory section has the same room temperature as the office section, the energy savings can be expressed by equation (8) according to equation (4) and (5):

$$\Delta\dot{Q} = 2(\dot{Q}_C - \dot{Q}_C^{'}) + (\dot{Q}_r + \dot{Q}_e + \dot{Q}_{eh} - \dot{Q}_i) - (\dot{Q}_r^{'} + \dot{Q}_e + \dot{Q}_{eh} - \dot{Q}_i^{'}) \qquad (8)$$

Equations (9) and (10) are deduced from equations (2) and (6):

$$\dot{Q}_r + \dot{Q}_e + \dot{Q}_{eh} - \dot{Q}_i = \dot{M}_i (h_r - h_{oa}) \qquad (9)$$

$$\dot{Q}_r^{'} + \dot{Q}_e + \dot{Q}_{eh} - \dot{Q}_i^{'} = \dot{M}_i^{'} (h_r - h_{oa}) \qquad (10)$$

Introducing equations (9) and (10) into equation (8), the potential energy savings is:

$$\Delta\dot{Q} = 2(\dot{Q}_C - \dot{Q}_C^{'}) + (\dot{M}_i - \dot{M}_i^{'})(h_r - h_{oa}) \qquad (11)$$

Where

$$\dot{Q}_C = \{\dot{M}_1(h_{m,1} - h_{c,1}), \dot{M}_1 C_p (T_{m,1} - T_{c,1}), 0\}^+ + \{\dot{M}_2(h_{m,2} - h_{c,2}), \dot{M}_2 C_p (T_{m,2} - T_{c,2}), 0\}^+ \qquad (12)$$

$$\dot{Q}_C^{'} = \{\dot{M}_1(h_{m,1}^{'} - h_{c,1}^{'}), \dot{M}_1 C_p (T_{m,1}^{'} - T_{c,1}^{'}), 0\}^+ + \{\dot{M}_2(h_{m,2}^{'} - h_{c,2}^{'}), \dot{M}_2 C_p (T_{m,2}^{'} - T_{c,2}^{'}), 0\}^+ \qquad (13)$$

$$\dot{M}_i = \dot{M}_1 \beta_{\min,1} + \dot{M}_2 \qquad (14)$$

$$\dot{M}_i^{'} = \dot{M}_r^{'} + \dot{M}_e + \dot{M}_2 \qquad (15)$$

$$h_{m,1} = (1 - \beta_{\min,1}) h_r + \beta_{\min,1} h_{oa} \qquad (16)$$

$$h_{m,1}^{'} = (1 - \beta_1) h_r + \beta_1 h_{oa} \qquad (17)$$

$$T_{m,1}^{'} = (1 - \beta_1) T_r + \beta_1 T_{oa} \qquad (18)$$

$$h_{m,2}^{'} = \begin{cases} (1-\beta_2) h_r + \beta_2 h_{oa} & Winter \\ \dfrac{\beta_2}{(\beta_2, \beta_c)^+} h_{oa} + \dfrac{(0, \beta_c - \beta_2)^+}{(\beta_2, \beta_c)^+} h_r & Summer \end{cases} \qquad (19)$$

$$T_{m,2}^{'} = \begin{cases} (1-\beta_2) T_r + \beta_2 T_{oa} & Winter \\ \dfrac{\beta_2}{(\beta_2, \beta_c)^+} T_{oa} + \dfrac{(0, \beta_c - \beta_2)^+}{(\beta_2, \beta_c)^+} T_r & Summer \end{cases} \qquad (20)$$

The cold deck air temperature of the laboratory section uses different set points during winter and summer for the humidity control. This impacts the outside air control schedule.

To eliminate the impact of building size on the saving values, the unit energy savings is defined as the ratio of the total energy savings to the total airflow rate to the office and laboratory sections.

$$\Delta\dot{q}_w = \Delta\dot{Q} / (\dot{M}_1 + \dot{M}_2) \qquad (21)$$

For winter months, the potential energy savings ($\Delta\dot{q}_w$) is expressed by equation (22):

$$
\begin{aligned}
\Delta\dot{q}_w = 2\varphi\{[(1-\beta_{\min,1})h_r + \beta_{\min,1}h_{oa} - h_{cw,1}], \\
C_p[(1-\beta_{\min,1})T_r + \beta_{\min,1}T_{oa} - T_{cw,1}], 0\}^+ + \\
2(1-\varphi)\{(h_{oa} - h_{cw,2}), C_p(T_{oa} - T_{cw,2}), 0\}^+ - \\
2\varphi\{[h_r - h'_{cw,1}) - \beta_1(h_r - h_{oa})], \\
C_p[(T_r - T'_{cw,1}) - \beta_1(T_r - T_{oa})], 0\}^+ - \\
2(1-\varphi)\{[(h_r - h'_{cw,2}) - \beta_2(h_r - h_{oa})], \\
C_p[(T_r - T'_{cw,2}) - \beta_2(T_r - T_{oa})], 0\}^+ - \\
\{[\gamma\varphi(h_r - h_{oa}) + \varphi(\beta_{\min,1} - \zeta)(h_r - h_{oa})], C_p \\
[\gamma\varphi(T_r - T_{oa}) + \varphi(\beta_{\min,1} - \zeta)(T_r - T_{oa})], 0\}^+
\end{aligned}
\tag{22}
$$

For summer months, the potential energy savings (Δq_s) is expressed by equation (23):

$$
\begin{aligned}
\Delta\dot{q}_s = 2\varphi\{[(1-\beta_{\min,1})h_r + \beta_{\min,1}h_{oa} - h_{cs,1}], \\
C_p[(1-\beta_{\min,1})T_r + \beta_{\min,1}T_{oa} - T_{cs,1}], 0\}^+ + \\
2(1-\varphi)\{(h_{oa} - h_{cs,2}), C_p(T_{oa} - T_{cs,2}), 0\}^+ - \\
2\varphi\{[(h_r - h'_{cs,1}) - \beta_1(h_r - h_{oa})], \\
C_p[(T_r - T'_{cs,1}) - \beta_1(T_r - T_{oa})], 0\}^+ - \\
2(1-\varphi)(\beta_2, \beta_c)^+ \\
\left\{\left[\frac{\beta_2}{(\beta_2, \beta_c)^+}h_{oa} + \frac{(0, \beta_c - \beta_2)^+}{(\beta_2, \beta_c)^+}h_r - h'_{cs,2}\right], C_p\right. \\
\left.\left[\frac{\beta_2}{(\beta_2, \beta_c)^+}T_{oa} + \frac{(0, \beta_c - \beta_2)^+}{(\beta_2, \beta_c)^+}T_r - T'_{cs,2}\right], 0\right\}^+ - \\
\{[\gamma\varphi(h_r - h_{oa}) + \varphi(\beta_{\min,1} - \zeta)(h_r - h_{oa})], C_p \\
[\gamma\varphi(T_r - T_{oa}) + T'_{cs,2}\varphi(\beta_{\min,1} - \zeta)(T_r - T_{oa}), 0]\}^+
\end{aligned}
\tag{23}
$$

Where

$$\gamma = \frac{\dot{M}_r}{\dot{M}_1} \tag{24}$$

$$\zeta = \frac{\dot{M}_e}{\dot{M}_1} \tag{25}$$

$$\varphi = \frac{\dot{M}_1}{\dot{M}_1 + \dot{M}_2} \tag{26}$$

$$\beta_c = \frac{T_r - T_{s,2}}{T_r - T_{c,2}} \tag{27}$$

$$\beta_2 = 1 - \frac{\varphi}{1-\varphi}(\beta_1 - \gamma - \zeta) \tag{28}$$

When heat recovery is included in both the base and LAHU systems, the condition of the air leaving the heat-recovery unit is nearly the same for both the base and LAHU systems. The potential energy savings can still be determined using equation (11). The condition of the outside air should be replaced by the condition of the air leaving the heat recovery unit.

ANALYSIS

The energy consumption was simulated for both base system and the LAHU system using the analytical models. Both the base and LAHU systems have the same cold deck set point for the office section and laboratory section during winter. During summer, the cold deck set point is 55°F for the office section. In winter, the cold deck set point is reset to 60°F to decrease reheat.

During summer, the cold deck set point of the base system is 55°F for the laboratory section. The cold deck set point of the LAHU system is set at 65°F. Because the laboratory section receives air from the office section, the higher set point will not create a humidity problem. The higher supply air temperature will decrease the re-heat significantly.

The room condition is assumed to be 75°F and 50% relative humidity. The actual room relative humidity conditions in laboratory section may be slightly different depending on the return air ratio. It is also assumed that common exhaust is 3% of the total supply airflow.

The potential energy savings depends on the total optimal outside air intake, the outside air intake to the office, and the outside air condition. The total optimal outside air intake can be expressed as the office airflow ratio (φ). If the office airflow ratio is 30%, the optimal outside air intake is 70% plus the common exhaust in the office section. The potential energy savings is calculated under different temperatures. The outside air relative humidity is assumed to be 50%. Under each temperature, the outside air intake fraction for the office section ranges from 10% to 100%.

Figure 3 presents the simulated contour line of potential thermal energy savings versus the outside air temperature (T_o) and office section outside air intake ratio (β_1). The contour line represents the same potential savings line. In this simulation, the office airflow ratio, φ, is 0.5. The energy savings is categorized into five regions.

Region I: part of winter operation. The outside air temperature is less than 50°F. The mixed air temperatures for both the office and laboratory sections are higher than or equal to the cold deck set points respectively. The low boundary corresponds to the critical office outside air intake ratio that makes the mixed air temperature equal its supply air temperature. The high boundary corresponds to the critical laboratory outside air intake ratio that makes the mixed air temperature equal its supply air temperature. LAHU systems consume no mechanical cooling energy. The potential thermal energy savings can be determined using equation (29):

$$\Delta \dot{Q} = 2\dot{Q}_C + (\dot{M}_i - \dot{M}_i^{'})(h_r - h_{oa}) \tag{29}$$

Both equations (29) and simulation results show that the potential thermal energy savings is independent of the office section outside air intake ratio. The base system needs mechanical cooling since it does not have an economizer. The heating energy savings decreases as outside air temperature increases. The cooling energy savings increases as the outside air temperature increases. The thermal energy savings (heating and cooling) increases as outside air temperature increases. When the outside air temperature increases from –20°F to 50°F, the thermal energy savings increases from 2.5 Btu/lbm to 3.4 Btu/lbm.

Region II: part of winter operation. The outside air temperature is lower than 50°F. The outside air intake ratios are lower than their critical values for both office and laboratory sections. In this region, mechanical cooling is required due to the outside air intake ratio. When the office section outside air intake ratio is lower than its critical value, potential energy savings decreases as the office section outside air intake ratio decreases. When the laboratory section outside air intake ratio is lower than its critical value, savings decreases as the office section outside air intake ratio increases. The savings increases as the outside air temperature increases. Under the same temperature, the savings in Region II is lower than the savings in Region I. The system should be controlled in Region I.

Region III: the outside air temperature is between 50°F and 55°F. It appears that the maximum thermal energy savings occurs when the office outside air intake is approximately 0.6. The savings decreases as the outside air intake ratios deviate from this value.

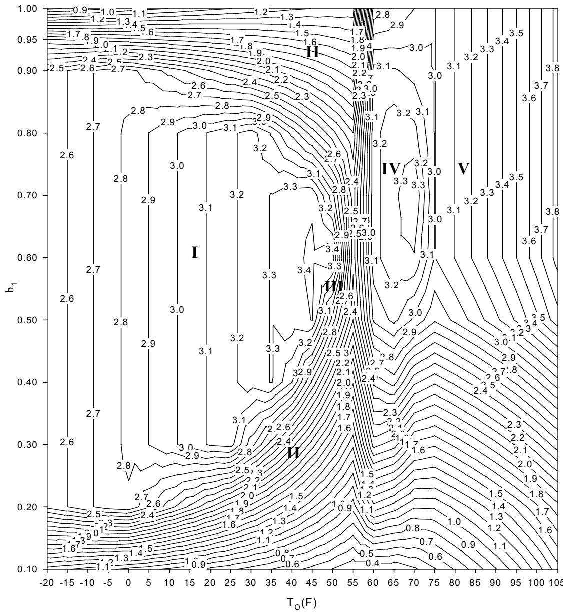

Figure 3. Potential Thermal Energy Savings Contour Line Versus Outside Air Temperature and Office Section Outside Air Intake Fraction (Office Total Air Flow Is 50% of the Total Building Airflow)

Region IV: the outside air temperature is between 55°F and 75°F. Outside air temperature is higher than cold deck air set point 55°F. Optimal energy savings ranges from 2.5 Btu/lbm to 3.4 Btu/lbm.
Under each outside temperature, the potential thermal energy savings increases as office section outside air intake ratio increases when it is lower than 0.68. With a low office section outside air intake ratio, the laboratory section supply air temperature is lower than required and reheat occurs. When the office section outside air intake ratio is increased, reheat energy consumption decreases and thermal energy savings increases. When the office section outside air intake ratio is higher than 0.68,the laboratory section cooling coil condensing energy consumption increases as office section outside air intake ratio increases. Thus, thermal energy savings decreases as office section outside air intake ratio increases.

Region V: the outside air temperature is higher than room temperature (summer). The optimal energy savings varies from 0.5 Btu/lbm to 3.8 Btu/lbm. Under each outside temperature, energy savings reaches the maximum value when office section outside air intake ratio is equal to or higher than 0.6. When the office section outside air intake ratio is 0.6 or higher, no reheat is required. The thermal energy savings is unchanged. When office section outside air intake ratio is lower than 0.6, the laboratory section supply air temperature is lower than the set point due to the requirement of humidity control, and thus reheat energy is consumed. Reheat energy consumption increases as the office section outside air intake ratio decreases. Thermal energy savings decrease as outside air intake ratio decreases. The optimal office section outside air intake ratio in Region V is 100%.

The simulation results show that the optimal energy savings varies from 2.5 Btu/lbm to 3.4 Btu/lbm for winter and from 3.0 Btu/lbm to 3.8 Btu/lbm for summer. The optimal office section outside air intake ratio varies from 0.65 to 1.00. It is important to point out that potential energy savings depends on outside air intake. Therefore, the outside air intake ratio must be properly controlled.

Figure 4 presents the simulated contour line of potential thermal energy savings versus the outside air temperature (T_o) and office section outside air intake ratio (β_1). The office section airflow ratio is 0.7. The optimal energy savings varies from 1.9 Btu/lbm to 2.9 Btu/lbm in summer and up to 4.3 Btu/lbm in winter.

It appears that the office section airflow ratio influences potential thermal energy savings, indoor air quality, and operational control of the system. More studies are required to draw conclusions.

CONCLUSIONS

The study shows that the LAHU system uses less thermal energy and provides better indoor air quality under all load and weather conditions. Thermal energy savings can be up to 3.8 Btu per pound supply airflow for typical laboratory buildings. The office section outside air intake ratio is often 100% during summer.

The developed models provide a simple method to analyze and calculate energy savings, avoiding treatment of space loads.

ACKNOWLEDGEMENT

The authors would like to express gratitude to Ms. Deborah Derrick for her editorial assistance.

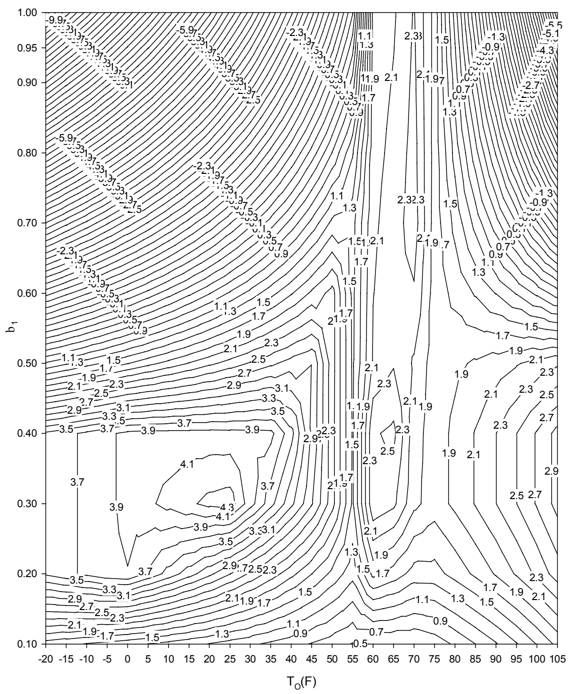

Figure 4. Potential Thermal Energy Savings Contour Line Versus Outside Air Temperature and Office Section Outside Air Intake Fraction (Office Total Air Flow Is 70% of the Total Building Airflow)

NOMENCLATURE

$\dot{Q}_i$, $\dot{Q}_i'$	total flow rate of energy carried by outside air in the base and LAHU systems, Btu/h
$\dot{Q}_h$, $\dot{Q}_h'$	total heating (preheat and reheat) supply flow rate to the control volume in the base and LAHU systems, Btu/h
$\dot{Q}_{hg}$	total internal heat gain flow rate of the control volume, Btu/h
$\dot{Q}_r$, $\dot{Q}_r'$	total energy flow rate carried by released air in the base and LAHU systems, Btu/h
$\dot{Q}_e$	total energy flow rate carried by exhaust air, Btu/h
$\dot{Q}_C$, $\dot{Q}_C'$	total cooling energy flow rate in the base and LAHU systems, Btu/h
$\dot{Q}_{eh}$	total energy flow rate that fume hood exhaust air carry in the lab section, Btu/h
$\dot{Q}_{env}$	total envelope load of the whole building, Btu/h

$\dot{Q}_{i,1}$, $\dot{Q}_{i,2}$ flow rate of energy carried by outside air into office and laboratory section in the base system, Btu/h

$\dot{Q}'_{i,1}$, $\dot{Q}'_{i,2}$ flow rate of energy carried by outside air into office and laboratory section in the LAHU system, Btu/h

$\dot{Q}_r$, $\dot{Q}'_r$ flow rate of energy carried by released air from office section in the base and LAHU systems, Btu/h

$\dot{Q}_{hg,1}$, $\dot{Q}_{hg,2}$ internal heat gain flow rate for office section and laboratory section, Btu/h

$\dot{Q}_{ph,1}$, $\dot{Q}_{ph,2}$ preheat flow rate to the office section and laboratory section in the base system, Btu/h

$\dot{Q}'_{ph,1}$, $\dot{Q}'_{ph,2}$ preheat flow rate to the office section and laboratory section in the LAHU system, Btu/h

$\dot{Q}_{Rh,1}$, $\dot{Q}_{Rh,2}$ reheat flow rate to the office section and laboratory section in the base system, Btu/h

$\dot{Q}'_{Rh,1}$, $\dot{Q}'_{Rh,2}$ reheat flow rate to the office section and laboratory section in the LAHU system, Btu/h

$\dot{Q}_{e,1}$, $\dot{Q}_{e,2}$ flow rate of energy carried by exhaust air from the office section and laboratory section, Btu/h

$\dot{Q}_{C,1}$, $\dot{Q}_{C,2}$ cooling energy flow rate for office section and laboratory section in the base system, Btu/h

$\dot{Q}'_{C,1}$, $\dot{Q}'_{C,2}$ cooling energy flow rate for office section and laboratory section in the LAHU system, Btu/h

$\dot{M}_i$, $\dot{M}'_i$ total outside air flow rate entering the control volume in the base and LAHU systems, lbm/h

$\dot{M}_r$, $\dot{M}'_r$ total released air flow rate from the control volume in the base and LAHU systems, lbm/h

$\dot{M}_e$ total exhaust air flow rate from the control volume, lbm/h

$\dot{M}_{eh}$ total fume hood exhaust air flow rate from the laboratory section, lbm/h

$\dot{M}_{e,1}$, $\dot{M}_{e,2}$ exhaust air flow rate from the office section and laboratory section, lbm/h

$\dot{M}_{i,1}$, $\dot{M}_{i,2}$ outside air flow rate entering office section and laboratory section in the base system, lbm/h

$\dot{M}'_{i,1}$, $\dot{M}'_{i,2}$ outside air flow rate entering the office section and laboratory section in the LAHU system, lbm/h

$\dot{W}_i$, $\dot{W}'_i$ total moisture intake flow rate to the control volume in the base and LAHU systems, lbm/lbma·h

$\dot{W}_p$ total moisture generation rate, lbm/lbma·h

$\dot{W}_{hu}$, $\dot{W}'_{hu}$ total moisture flow rate generated by humidifier in the base and LAHU systems, lbm/lbma·h

$\dot{W}_C$, $\dot{W}'_C$ rate of moisture removal by cooling coil in the base and LAHU systems, lbm/lbma·h

$\dot{W}_e$ moisture flow rate in exhaust air, lbm/lbma·h

$\dot{W}_{eh}$ moisture flow rate in fume hood exhaust air, lbm/lbma·h

h_r, h_{oa} room and outside air enthalpy, Btu/lbm

$h_{cw,1}$, $h_{cw,2}$ office section and laboratory section cold deck air enthalpy in winter, Btu/lm

$h_{cs,1}$, $h_{cs,2}$ office section and laboratory section cold deck air enthalpy in summer, Btu/lbm

$h_{s,2}$ laboratory section supply air enthalpy, Btu/lbm

T_r, T_{oa} room and outside air temperature, °F

$T_{cw,1}$, $T_{cw,2}$ office section and laboratory section cold deck air temperature in winter, °F

$T_{cs,1}$, $T_{cs,2}$ office section and laboratory section cold deck air temperature in summer, °F

$T_{s,2}$ laboratory section supply air temperature, °F

$h'_{m,1}$, $h'_{m,2}$ mixed air enthalpy for office section and laboratory section in the LAHU system, Btu/lbm

$h_{m,1}$ mixed air enthalpy for office section in the base system, Btu/lbm

β_1, β_2 outside air intake ratio for office section and laboratory section

$\beta_{\min,1}$ minimum outside air intake ratio for office section IAQ requirement

β_C critical outside air intake required to make the mixed air temperature after the cooling coil equal the supply air temperature for laboratory section

φ office section supply airflow rate to the total building supply airflow rate

REFERENCES

Bard, E. M. 1994. "Energy Efficient HVAC System Features Thermal Storage And Heat Recovery." *ASHRAE Journal* V. 36.

Bard, E. M.1995. "Laboratory Integrates VAV Fume Hood Controls With Central Building Automation System." *ASHRAE Journal* V. 37.

Barker, P. L. 1994. "Biomedical Laboratory Uses VAV Fume Hoods And Heat Recovery to Save Costs." *ASHRAE Journal* V. 36.

Davis, S. J. and R. Benjamin. 1987. "VAV With Fume Hood Exhaust System." *Heating/Piping/Air Conditioning* V. 59.

Neuman, V.A. and W.H. Rousseau. 1986. "VAV For Laboratory Hoods — Design And Costs." *ASHRAE Transactions* Vol. 92, Part 1A.